

Improve exchanger operability and efficiency through tube-side enhancement

The authors' company has been serving the process industry for more than 35 yr to improve tube-side heat transfer in tubular heat exchangers. Its main application area is the use of a proprietary tube insert technology^a in new equipment design for exchangers with laminar tube-side flow, where considerable size reductions are achievable due to the increased tube-side performance. An increased focus exists for the use of this technology in heat exchanger revamps. Aside from the obvious potential of increased duty and, therefore, increased exchanger efficiency, applying this technology can also often provide improved operability.

Revamp considerations. The main driving forces to undertake heat exchanger revamps on existing units can be divided as follows:

- Increased overall heat transfer coefficient
- Improved fluid distribution
- More stable operation and longer run times.

The tube insert technology enhancement affects only the thermal tube-side performance; whereas, for an exchanger revamp, the impact on overall performance must be determined. The heat duty of a heat exchanger can be approximated in Eq. 1:

$$Q = U \times A \times \text{EMTD} \quad (1)$$

The increase in overall coefficient U determines the efficiency of the revamp. Since this coefficient is also a function of the wall and fouling resistance and the tube outside coefficient, the impact of the revamp depends very much on those variables, shown in Eq. 2:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + \text{Fouling resistance} + \text{Wall resistance} \quad (2)$$

In the revamp example in FIG. 1, the tube-side heat transfer increased almost nine-fold, from 160 W/m²K to 1,400 W/m²K, at an identical pressure drop of 1 bar. The impact on the overall coefficient and, therefore, on the effectiveness of the revamp, depends on the other heat transfer resistances in Eq. 2.

This is demonstrated in FIG. 2 by considering different tube outside coefficients (h_o). It is evident that with a higher tube outside coefficient, the revamp becomes more tube-side controlled and, therefore, more efficient. For a heavily tube-side-controlled scenario, the overall multiplier approaches the increase in tube-side coefficient only. The tube insert technol-

ogy revamps operations to reach the highest efficiency in the case of heavily tube-side-controlled operation.

Benefits of increased heat transfer coefficient. Eq. 1 can be used to discuss the different revamp scenarios that focus on the increase in tube-side heat transfer and, consequently, in the increase of overall coefficient. An example for the retrofit of a viscous oil heater demonstrates the different possible revamp scenarios. In the base case, before revamping, the oil is heated from 120°C to 163°C with condensing steam (20 bar) on

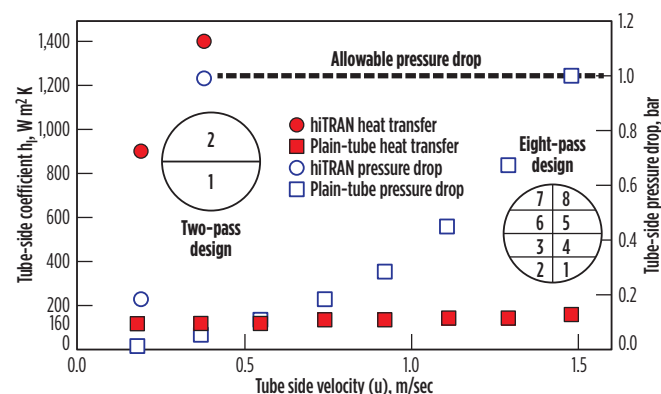


FIG. 1. Heat transfer and pressure drop as a function of tube velocity (pass arrangement) for a typical proprietary tube insert technology^a design.

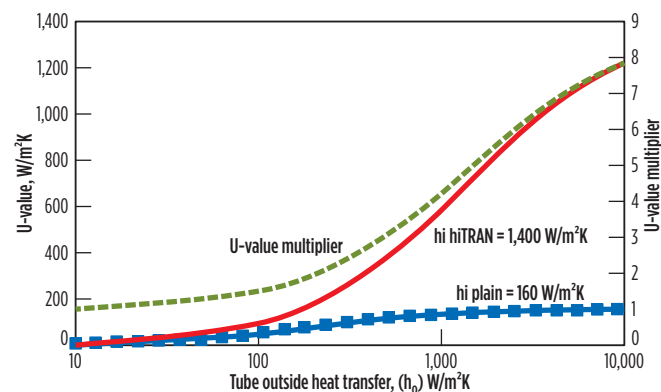


FIG. 2. Change of U-value and U-value multiplier, which increases tube outside coefficient (h_o).

the shell side. Due to the viscosity of the viscous oil between 7cP and 17cP, the flow in the eight-pass AES heat exchanger (236 tubes 6 m × 26.4 mm × 2.11 mm) ranges from laminar to transitional, resulting in a low tube-side heat transfer coefficient. The constraints for the different revamp scenarios are a constant heat transfer area *A* and also the maximum allowable pressure drop of 1 bar.

The summary of base conditions is shown in **TABLE 1**. Where the exchanger operates before revamp, as in this case, at the tube-side pressure drop limit of 1 bar, the number of tube passes must be reduced to one to reduce the flow path length. This is a contributing factor to the tube-side pressure drop.

This option is feasible only if the exchanger is designed as a multi-pass unit. To keep the piping arrangement for the exchanger unchanged, in general, only an even-to-even modification is considered. After removal of pass partition plates, the flow velocity is reduced accordingly (**FIG. 3**). The different possible revamp scenarios are calculated with a thermal process design and simulation software^b comprising a proprietary design and selection program software^c plug-in and later summarized in **TABLE 1**.

The impact on the shell-side performance must be considered; increased tube-side duties may result in change in shell-side outlet temperatures, which affect the effective mean tem-

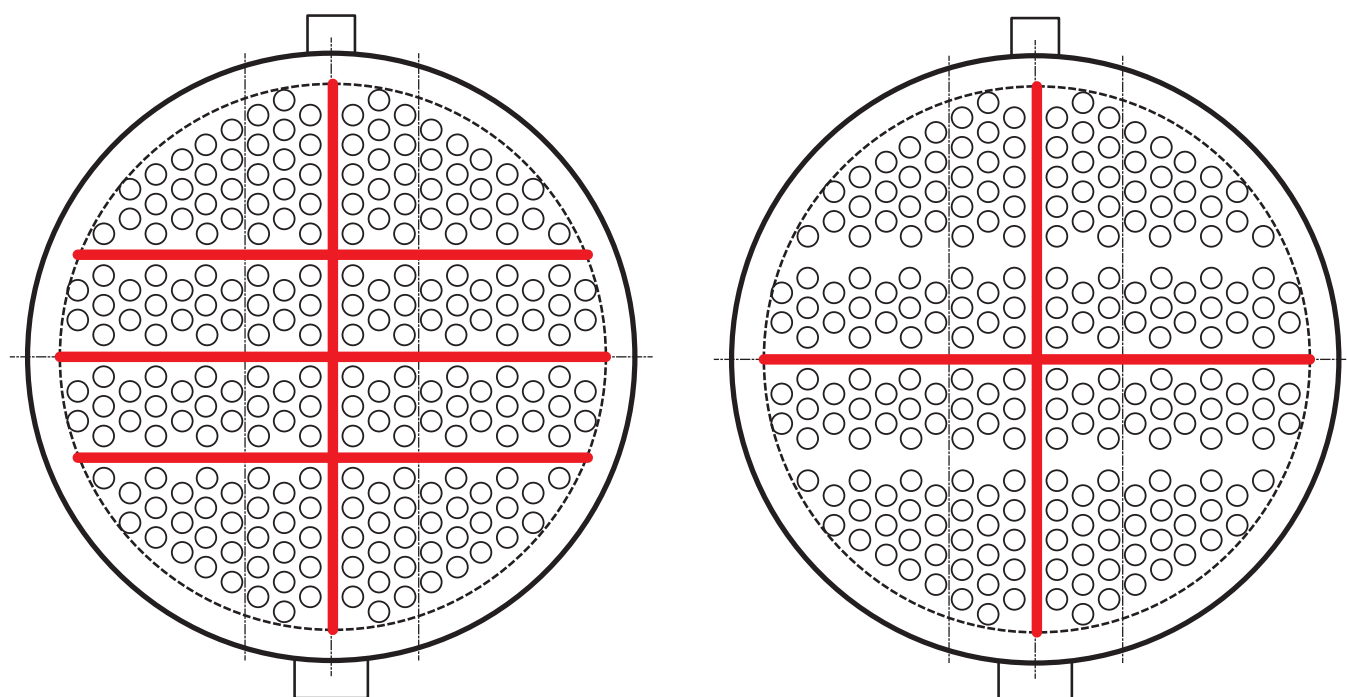


FIG. 3. Pass arrangement reduced from eight to two passes, shown in HTRI for the example case.

TABLE 1. Summary of the different revamp scenarios discussed (red values highlight main achievements)

	Base case	4.1.1 Increased throughput Two-pass	4.1.1 Increased throughput One-pass	4.1.2 Increased outlet temp	4.1.3 Reduced steam pressure
Shell side					
Condensation pressure, bar	20	20	20	20	5.2
Condensation temperature, °C	212	212	212	212	153
Tube side					
Number of passes	8	2	1	2	2
Flowrate, kg/sec	12	32	85	12	12
Temperature out, °C	163	180	151	211	151
Calculated pressure drop, bar	1	1	1	0.85	1
Velocity/shear equivalent, m/sec	1.3/1.3	0.9/1.4	1.15/2	0.33/0.9	0.33/1.1
Reynolds Number, -	920/2,650	650/2,800	860/2,200	250/1,700	240/650
Heat transfer, W/m ² K	172	1,016	1,354	1,405	1,001
Duty, MW	1.3	5.4	9.1	2.8	1.3

perature difference (EMTD). In this case, the use of condensing steam means that the overall EMTD is largely unaffected.

Increased throughput. The purpose of a revamp is often to increase plant throughput; here, heat exchangers can be the bottleneck since they are designed for specified mass flow. In the case of tube-side-controlled operation, as discussed here, the increase in tube-side heat transfer translates to a much-improved overall coefficient (U). In this case, with condensing steam on the shell side, the temperature levels for both streams are kept constant, and the potential increase in mass flow will be proportional to the increase in overall heat transfer.

Since both the increased mass flow and the use of enhancement inside the tubes will add pressure drop, the number of passes must be reduced to stay within the allowable pressure drop, as explained before. In this case, a reduction from eight to two passes will allow an increase in the friction factor of 64 times for the modified arrangement, without exceeding the pressure drop (Eq. 3).

$$\left[\frac{N_{p(plain)}}{N_{p(hiTRAN)}} \right]^3 \times f_{plain} \geq f_{hiTRAN} \quad (3)$$

The mass flow can be increased from 12 kg/sec to 32 kg/sec within the allowable pressure drop of 1 bar. The increased heat transfer from the design and selection program software allows increased heating of the fluid beyond the required 151°C. This suggests that a considerable further increase in flowrate is possible, which can be achieved by a change to a one-pass arrangement, requiring a modification to the piping and headers. As a result, the flowrate can be increased by the factor of seven to approximately 85 kg/sec within the allowable pressure drop.

Increased/reduced outlet temperatures. In other revamp situations, the mass flow through the exchanger remains unchanged, and the aim is to achieve a higher temperature change of the process stream. An example is product air coolers, where the outlet temperature for storage can be reduced. In feed/effluent exchangers, the increased heat transfer equates to higher heat recovery rates and can be used to reduce the load on the fired heater. Again, the level of improvement possible depends on the tube-side flow condition and how much the overall coefficient is affected by the change in tube-side performance.

In this example, the pass arrangement of the exchanger must again be modified to operate within the allowable pressure drop. As a result of the much-improved tube-side heat transfer coefficient, the outlet temperature increases from 151°C to 211°C, as shown in **TABLE 1**. The outlet temperature pinches with the steam temperature of 212°C at this level. Such revamp option temperature pinches must be considered. An additional benefit is an additional capacity to allow for fouling margins.

Reduced temperature driving force. The aim of a revamp can also be to run an exchanger for the same duty with a reduced temperature driving force. Examples include increasing ambient air temperatures beyond the design values, leading

to an increasing number of underperforming air coolers. In those situations, an increased tube-side coefficient and, consequently, higher overall heat transfer can compensate for a loss in EMTD, as seen in Eq. 1.

A different scenario is demonstrated with the viscous steam heater example. The increased tube-side coefficient can be used to operate the exchanger with significantly reduced condensing steam pressure. In this case, the steam pressure can be reduced from 20 bar to 5.2 bar. This often means that lower-quality steam can be used. Alternatively, due to the lower temperature level, the process can be switched from steam-heated to a heat transfer fluid-heated process with associated benefits. Apart from the cost reduction, a reduced temperature driving force will lead to reduced wall temperatures. For liquids sensitive to temperature-driven chemical reaction fouling, the drop-in wall temperatures can have a dramatic effect on fouling behavior.

Takeaway. Detailed information of the potential benefits when using the proprietary tube insert enhancement technology has been provided. Those benefits are divided into operational process improvements, increased exchanger duty and improvements that affect a better fluid distribution in the equipment. A substantial duty increase is possible where the tube-side heat transfer is the controlling heat transfer resistance in the unit. It is also evident that minor changes to the pass partitions in the header opens considerable possibilities for revamp in existing units. The aim is to encourage the plant operator and thermal design engineer to understand the opportunities to revamp with this technology. **HP**

NOTES

^a CALGAVIN's hiTRAN tube insert technology

^b HTRI Xchanger suite

^c CALGAVIN'S hiTRAN.SP



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the company's state-of-the-art, in-house research facilities. He is also responsible for cooperation with various external research institutions in the UK and Europe. Based on these research activities, he took a leading role in developing the company's design and selection program software^c, which was first launched in 2007 as a standalone version. He was then subsequently responsible for the integration with Aspen EDR and HTRI Xchanger Suite products. Dr. Drögemüller has also authored various scientific papers and magazine articles, and represents CALGAVIN at international meetings and with industrial clients in aspects of heat transfer engineering and exchanger debottlenecking.



PETER ELLERBY joined CALGAVIN in 1996 as an Engineering Manager with overall responsibility for thermal design, engineering and technical proposals. During his time with CALGAVIN, Mr. Ellerby has overseen many new and retrofit projects worldwide, from conceptual heat exchanger design to startup. He also made a significant contribution to the development and use of the company's proprietary design and selection program software^c tool and has pioneered the application of enhancement in new applications, including two-phase systems. He has represented CALGAVIN on technical committees, at conferences and exhibitions, and has authored a range of papers on specialist aspects of heat transfer and heat exchanger design. He earned a BS degree in chemical engineering. Previously, Mr. Ellerby worked for Graham Manufacturing, Tioxide Europe Ltd. and Heatric.